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Model for assessing the road train stability movement

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Abstract. Road trains (RT) are the main transport in agriculture. However, their accident rate is quite high. In the vehicles movement research their design characteristics and operating modes are mainly improved. The main way to reduce the accident rate remains the speed limit. The authors proposed methods (confirmed by the issued patents) for ensuring the stability of the RT movement by changing the design of trailer devices. The purpose of this work is to form a mathematical model for assessing the stability of the RT movement in the presence of side effects (modeling of sharp turns, impacts). Methods of analysis and synthesis, mathematical analysis, and vector geometry were applied. The process of RT yawing on the highway is split into a number of stages, for which the laws of changing its geometry and kinematics are formulated. Side effect leads to increase in the RT impulse (lateral oscillations increase the speed), or to a violation of stability (the movement amplitude goes beyond the highway or tipping). This model can be used to analyze the stability of the RT movement and evaluate methods to influence it in order to reduce the probability of accidents.

1. Introductions

The main method of cargo transportation in the agro-industrial complex remains the use of motor vehicles. Therefore, it is extremely important not only to improve their efficiency (for example, improving maneuverability), but also to reduce accidents on the roads (for example, by ensuring the stability of their movement on the highway).

Every tenth accident (road traffic accident) in the Russian Federation occurs with the participation of trucks and road trains (RT) [1]. Given the fact that only in 2019, the number of accidents in Russia amounted to 147738, number of affected people – 189671, including – 15158 fatal accidents [2], the study of problems of the stability of motion of RT is quite relevant.

Recent studies of the movement of trucks and tractors with trailers on the road, mainly consider the optimization of the design characteristics of the suspension mechanisms, control, braking (impact on the stability of the vehicle tire deformation [3], the parameters of the suspension mechanism [4], application of motion controllers [5]), as well as improvement of operational modes of RT [6-8]. For example, in [6] GPS antennas are used for traffic control, in [7] models of torque vectorization of electric vehicles are analyzed, in [8] an algorithm for evaluating critical vehicle speeds depending on



its turning radius and tire sizes is presented, in [9] a model for analyzing the dynamics of a freight cars movement depending on the parameters of the highway and wheel wear is presented, in [10] a mechanism for improving the stability of the car using front wheel control controllers is considered, in [11] we study the process of longitudinal control when Parking a vehicle, taking into account the lateral forces of the front tires, at large steering angles, in [12] we present a model for assessing the impact of road parameters and conditions on the speed of vehicles, in [13] the stability of the vehicle is evaluated based on the analysis of its dynamic characteristics (weight, driving maneuverability) and road conditions, in [14] the controllability and stability of a car with a trailer is evaluated depending on the vertical load, in the work [15], the lateral stability of the vehicle is modeled depending on the radius of the bend of the road route, the roll and skid of the car. However, with all the variety of proposed design and technological solutions, the main method to ensure the stability of the RT movement on the highway remains the speed limit, which significantly limits the effectiveness of its use. Previously, we proposed the method for ensuring the RT movement stability by changing the design of trailer devices [16], the relevance and practical significance of which is confirmed by a number of patents issued. However, to quantify the effectiveness of the proposed method, a mathematical apparatus is required that would allow both a deep analysis of the process for ensuring the RT movement stability, and a correct comparison of calculated and experimental data.

The purpose of this work is to create a mathematical model for analyzing and evaluating the parameters of the RT movement stability, which would allow predicting the behavior of the RT during its movement on the road, in the presence of lateral external effects (imitation of sharp turns, displacement of cargo in the vehicle, impacts).

Such a model will allow for a more accurate analysis of the RT movement on highways, identify critical modes of function and justify ways and directions to improve the stability and steerability of vehicles in general.

2. Methodology

To solve this problem, the methods of analysis and synthesis of mechanisms, mathematical analysis, vector geometry, and the similarity method were used.

The RT composition is shown in the figure 1. In General, RT can be represented as a three-link mechanism (figure 2) with two movable joints. Moreover, the elements P and TS can be taken rigidly fixed (relative to the line O of the RT movement), and the element D – with ability to rotate relative to the O axis.

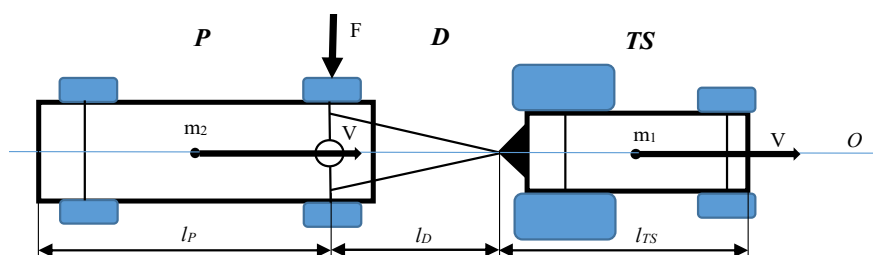


Figure 1. Composition of RT: TS – transporting vehicle, P – trailer with a Pivot Platform (PP), D – tow bar, F – external influence, l_{TS} , l_D , l_P – length of the transport vehicle, tow bar and trailer, respectively.

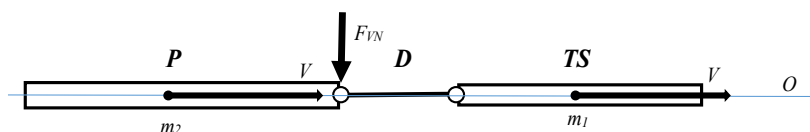


Figure 2. Diagram of the three-link RT mechanism.

The main task of ensuring RT stability movement along a straight line route is to consider the possibility of maintaining the specified, optimal trajectory of its movement, especially in case of external destabilizing influence. With RT curvilinear motion (rotation, changing lane, impacts) the additional complicating factor will be the inertia forces of the TS and P elements, increasing the possibility of lateral displacement (skidding) of the RT elements relative to its trajectory.

When forming a model for assessing the stability of behavior (moving along the highway) RT in the case of external influence, you can use the law of conservation of impulse of a closed system. To analyze the behavior of the RT mechanism, in this case, the basic principles of forming mechanisms may be used. The stability of the behavior (movement) of the RT on the highway should be assessed by the probability of returning to the original trajectory of movement after external influence.

3. Results and discussion

Changing the pulse of a closed system is possible only due to the application of external forces, and not only the volume and direction of the external impact of the F_{VN} , but also its duration are important:

$$\frac{d\vec{p}}{dt} = \frac{d(m\vec{V})}{dt} = \vec{F}_{VN} \quad (1)$$

where $d\vec{p}$ – system impulse change; dt – external impact duration.

Then the mechanism of external influence on the moving RT must confirm the implementation of this law:

$$m_{TS}\vec{V}_{i_{TS}} + m_P\vec{V}_{i_P} = \text{const} = m_{TS}\vec{V}_{i+1_{TS}} + m_P\vec{V}_{i+1_P} + \vec{F}_{VN}t \quad (2)$$

where i – is the stages (time) of the RT movement.

The change in the position of the RT elements due to external influence, divided into several characteristic stages, is shown in figures 3-8.

Stage 1. External exposure to F_{VN} increases the total RT impulse. Moreover, the impulse of the last element RT – P will be changed, because trailer PP fixed free and has the ability to quite easily move against the elements TS and P. The change in the impulse of the Vehicle element can be ignored, as well as the resistance to movement of the element – D. We will also consider the friction forces on the roadbed insignificant: for all the wheels of the first (TS) and third (P) elements of the RT mechanism.

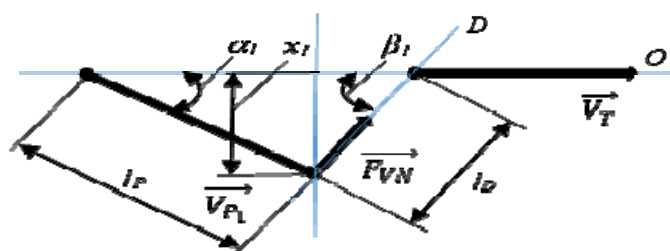


Figure 3. Position of RT elements at the end of Stage 1.

The system's impulse conservation law at the end of the 1 Stage can be formulated as:

$$m_{TS}\vec{V}_{TS} + m_P\vec{V}_P + \vec{F}_{VN}t = m_{TS}\vec{V}_{TS} + m_P\vec{V}_{P_1} = \text{const} \quad (3)$$

The speed of the Element P, at the end of the 1 Stage, can be defined as:

$$\vec{V}_{P_1} = \frac{m_P\vec{V}_P + \vec{F}_{VN}t}{m_P} \quad (4)$$

$$\vec{V}_{P_1} = \frac{F_{VN}t}{m_p \sin \beta_1} \quad (5)$$

The lateral component of the element P speed (the speed of the sidewise skidding):

$$V_{P_s} = \frac{F_{VN}t}{m_p} \quad (6)$$

The value of the lateral offset (skidding) of the P element will be:

$$x_1 = V_{P_s}t = \frac{F_{VN}t^2}{m_p} = l_D \sin \alpha_1 = l_p \sin \beta_1 \quad (7)$$

The angles of deviation of the tow bar (element D) and trailer (element P) axes from the direction of travel at the end of the 1 Stage will be, respectively:

$$\alpha_1 = \arcsin\left(\frac{x_1}{l_D}\right) = \frac{l_p}{l_D} \sin \beta_1 \quad (8)$$

$$\beta_1 = \arcsin\left(\frac{x_1}{l_p}\right) = \frac{l_D}{l_p} \sin \alpha_1 \quad (9)$$

Stage 2. Spatial inconsistency of the vectors V_{P_1} and V_T of the P and TS elements leads to the appearance of a stabilizing internal force F_{ST} , which tries to restore the original position of the P and TS elements of the mechanism (figure 3). The return of the P element to its original position (the velocity vector V_{P_2} coincides with the axis of motion O) is due to the displacement (rotation) of the TS element of the mechanism relative to the O axis.

Expression (2) will take the form:

$$m_T \vec{V}_T + m_p \vec{V}_{P_2} = m_T \vec{V}_{T_2} + m_p \vec{V}_{P_2} = \text{const} \quad (10)$$

The movement of the RT elements will be determined by the movement of the D element, whose movement (figure 4), in turn, will be determined by the pulse acting on it (the shifting force V_{sD} , equal to the sum of the projections of the pulses of the elements P and TS on the D axis):

$$\vec{p}_D = m_T V_T \cos \alpha_1 + m_p V_{P_1} \sin(90^\circ - \alpha_1 - \beta_1) \quad (11)$$

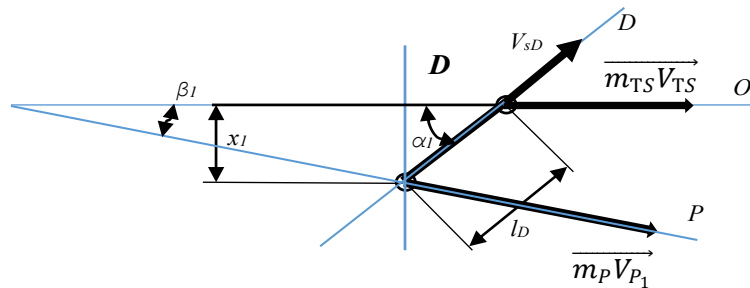


Figure 4. Forces acting on element D at the beginning of Stage 2.

This Stage ends when the P element is combined with the O axis. The duration of the return of the P element to the O axis (the time of the stage) will be determined by the projection of the sum of the pulses of the P and TS elements on the D axis:

$$\vec{p}_D = (m_P V_{P_1} \sin(90^\circ - \alpha_1 - \beta_1) + m_{TS} V_{TS} \cos \alpha_1) t_2 = (m_P + m_{TS}) \vec{V}_{sD} = \frac{(m_P + m_{TS}) l_D}{t_2} \quad (12)$$

$$t_2 = \sqrt{\frac{(m_P + m_{TS}) l_D}{m_P V_{P_1} \sin(90^\circ - \alpha_1 - \beta_1) + m_{TS} V_{TS} \cos \alpha_1}} \quad (13)$$

The value of the lateral displacement (skidding) of TS and P elements will be:

$$x_2 = x_{TS} = l_D \sin \alpha_1 = x_1 \quad (14)$$

The speed V_{sD} of the element D along the direction of its movement (D axis):

$$\vec{V}_{sD} = \frac{l_D}{t_2} \quad (15)$$

and its projection on a line perpendicular to the O axis will determine the speed of sidewise skidding (shift) of the P element (as well as the TS):

$$V_{Ps_2} = V_{sD} \sin \alpha_1 = \frac{l_D}{t_2} \sin \alpha_1 = \frac{x_2}{t_2} \quad (16)$$

The angle of deviation of the TS element axis from the O axis will be:

$$\gamma_2 = \arcsin\left(\frac{x_2}{TS}\right) \quad (17)$$

Scalar values of RT element velocities at the end of Stage 2:

$$V_{TS_2} = \frac{(m_P V_{P_1} \sin \beta_1)}{(m_{TS} \sin \gamma_2)} \quad (18)$$

Stage 3. Element P, having taken its original position (along the axis O, figure 5), under the action of the inertia force F_{in} , takes a position (rotates), as shown in figure 6. In this case, the inertia forces act only on the element P, which (due to the presence of the PP) has the ability to shift relative to the axis O. The position of the TS element at this Stage can be assumed unchanged.

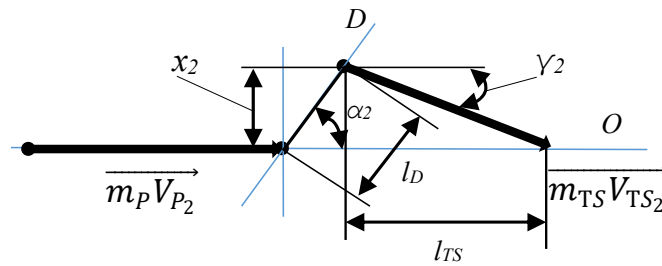


Figure 5. Position of RT elements at the beginning of Stage 3.

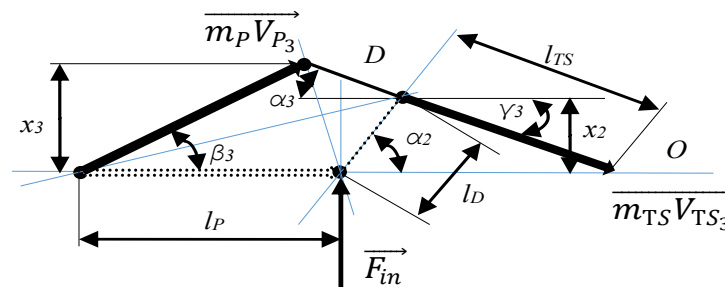


Figure 6. Position of RT elements at the end of Stage 3.

Thus, the inertia force F_{in} can be considered as perturbing (generating the oscillatory process of RT elements displacement relative to the O axis):

$$F_{in} = m_p \alpha_2 = m_p \frac{2x_2}{t_2^2} \quad (19)$$

where a_2 – is the acceleration acquired by element P at the end of Stage 2.

The duration of the stage will be determined by the end of the inertia forces. When evaluating the rotation of the element P relative to the axis O, based on the condition of the triangles symmetry formed by the arrangement of the elements P and D, the following dependencies were obtained:

$$\beta_3 = 2\arctg\left(\frac{l_D \sin \alpha_2}{l_P + l_D \cos \alpha_2}\right) \quad (20)$$

$$\alpha_3 = \arccos\left(\frac{l_p + l_D \cos \alpha_2 - l_p \cos \beta_3}{l_D}\right) \quad (21)$$

$$\gamma_3 = \gamma_2 \quad (22)$$

The value of the lateral displacement x_3 of the element P at the end of the 3 Stage:

$$x_3 = l_p \sin \beta_3 \quad (23)$$

Time of action of inertia forces on element P (time of the 3 Stage):

$$t_3 = \sqrt{\frac{2x_3}{\alpha_3}} \quad (24)$$

Stage 4. Spatial inconsistency of the impulse vectors of the elements P and TS acting on the element D leads to its rotation relative to the RT mechanism. In this case, element D, turning clockwise, at the beginning of the 4 Stage, is combined with the axis of the element TS, contributing to the return of the impulse vector of the element TS to the axis of movement O (figure 7), and then, continuing its turn, – accelerates the process of this return. At the end of the 4 Stage, the elements of the RT mechanism assume a position that is mirror-symmetrical to their position at the beginning of the 2 Stage (figure 8).

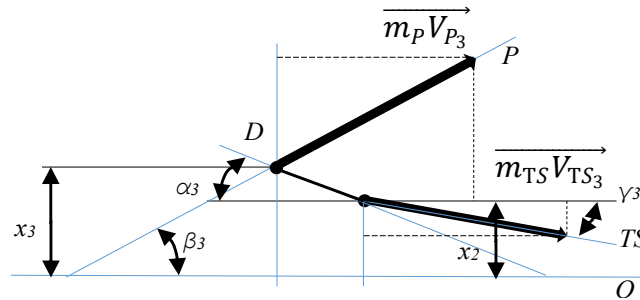


Figure 7. Forces acting on element D at the beginning of Stage 4.

The condition for preserving the impulse of the movement of the RT mechanism, at the end of the 4 Stage, will take the form:

$$m_{TS} \vec{V}_{TS} + m_P \vec{V}_P = m_{TS} \vec{V}_{TS_2} + m_P \vec{V}_{P_2} = m_{TS} \vec{V}_{TS} + m_P \vec{V}_{P_4} = \text{const} \quad (25)$$

Time of the 4 Stage:

$$t_4 = \frac{V_{TS_4} \sin \gamma_3}{x_2} \quad (26)$$

The value of the lateral displacement x_4 of element P at the end of the 4 Stage:

$$x_4 = x_3 + V_{P_4} \cos \beta_3 \quad (27)$$

The angle of deviation of the element P from the axis O will be defined as:

$$\beta_4 = \arcsin \left(\frac{x_4}{l_P} \right) \quad (28)$$

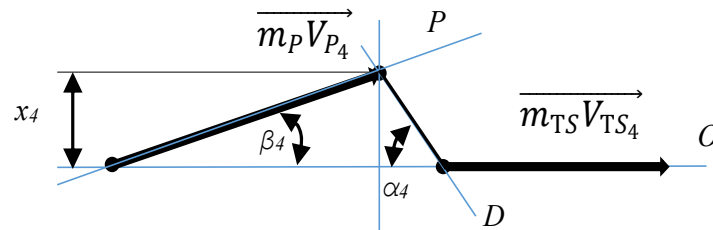


Figure 8. Position of the RT mechanism at the end of Stage 4.

In the future, Stage 2-4 are repeated. Thus, this process can be classified as oscillation categories. The change in the lateral displacement value (skidding) of elements x , as well as the displacement angles (α , β , or γ), can be considered as physical quantities of the oscillatory process. Their decrease (with each new cycle) will characterize the process as decaying, and vice versa.

For each stage, critical modes (values of skid (lateral displacement) x and angles of displacement (α , β , γ) of the elements of the road train can be set. Thus, according to calculations, the maximum value of the lateral impulse F_{VN} should not be more than 0.5-1.0 % of the total impulse of the road train. Otherwise, the road train will go beyond the transport corridor (go off the road) and it will be pointless to talk about stability. The critical values of the angle of rotation of the trailer drawbar under side impact are 30-88° (for m_{TS} =600-3000 kg, m_P = 300-5000 kg, V =3-20 m/s, l_D = 2.0 m, l_P =6.0 m) and increase with increasing total system momentum. This model also allows us to quantify (and practically test) the effectiveness of measures to improve the stability of the road train, proposed in [16].

4. Conclusion

The proposed model allows you to divide the entire process of movement of elements of a road train (in the presence of external influence) into several characteristic stages. For each stage, characteristic (boundary) modes of movement of the road train (values of skid (lateral displacement) x and displacement angles (α , β , γ) of elements, the maximum permissible speed) can be established (and verified by practical experiments).

Giving an external impulse (impact) to a moving road train leads either to a violation of its movement stability (fluctuations in the movement amplitude go beyond the safe corridor), or to a change in the total impulse of the entire system. In the second case, either the speed of the road train changes, or the vibrations are extinguished by internal deformations and forces (friction, inertia). Changes in the design of the trailer device proposed in [16] allow the use of internal forces and deformations to dampen fluctuations in the movement of the road train on the highway, which will increase the minimum speed and safety of its movement.

These technical solutions, confirmed by the materials of several patents and by laboratory tests, allow you to get a significant economic effect in the agro-industrial complex by reducing traffic accidents.

References

- [1] The most frequent accidents in Russia, available at: <https://journal.tinkoff.ru/stat-dtp/> [Accessed 21 January 2020]
- [2] Road safety indicators, available at: <http://stat.gibdd.ru/> [Accessed 21 January 2020]
- [3] Simniceanu L 2019 The study of the car's stability using a simplified model *IOP Conf. Ser.-Mat. Sci.* **568** 012053 <https://doi.org/10.1088/1757-899X/568/1/012053>
- [4] Fedotov A I, Tikhov-Tinnikov D A, Ovchinnikova N I and Lysenko A V 2017 Simulation of car movement along circular path *IOP C. Ser. Earth Env.* **87** 082018 <https://doi.org/10.1088/1755-1315/87/8/082018>
- [5] Prochowski L, Pusty T, Gidlewski M and Jemioł L 2018 Experimental studies of the car-trailer system when passing by a suddenly appearing obstacle in the aspect of active safety of autonomous vehicles *IOP Conf. Ser.-Mat. Sci.* **421** 032024 <https://doi.org/10.1088/1757-899X/421/3/032024>
- [6] Kayacan E, Kayacan E, Ramon H and Saeys W 2014 Distributed nonlinear model predictive control of an autonomous tractor-trailer system *Mechatronics* **24** 926 <https://doi.org/10.1016/j.mechatronics.2014.03.007>
- [7] Weigel-Milleret K and Grzegożek W 2018 Modelling and simulation of narrow car dynamic *IOP Conf. Ser.-Mat. Sci.* **421** 022038 <https://doi.org/10.1088/1757-899X/421/2/022038>
- [8] Taneva S, Ambarev K and Katsov D 2019 Study on the critical speeds of car's movement with low and ultra-low profile tires in a turn *IOP Conf. Ser.-Mat. Sci.* **618** 012065 <https://doi.org/10.1088/1757-899X/618/1/012065>
- [9] Tarmaev A A, Petrov G I and Filippov V N 2019 Analysis of freight cars wheels wear based on mathematical modeling of the dynamics of their movement *J. Phys. Conf. Ser.* **1172** 012087 <https://doi.org/10.1088/1742-6596/1172/1/012087>
- [10] Mayurika Chatterjee, Mangesh Kale and Bhalchandra Chaudhari 2019 A dynamic stability control for electric narrow tilting three wheeled vehicle using integrated multivariable controller *Transport. Res. D. E.* **66** 58 <https://doi.org/10.1016/j.trd.2017.08.006>
- [11] Martina Joševski, Alexander Katriniok, Andreas Riek and Dirk Abel 2017 Disturbance Estimation for Longitudinal Vehicle Dynamics Control at Low Speeds *IFAC-PapersOnLine* **50** 987 <https://doi.org/10.1016/j.ifacol.2017.08.204>
- [12] Kozlov V G, Skrypnikov A V, Samcov V V, Levushkin D M, Nikitin A A and Zaikin A N 2019 Mathematical models to determine the influence of road parameters and conditions on vehicular speed *J. Phys. Conf. Ser.* **1333** 032041 <https://doi.org/10.1088/1742-6596/1333/3/032041>

- [13] Abdullah M A, Jamil J F and Salim M A 2015 Dynamic performances analysis of a real vehicle driving *IOP Conf. Ser.-Mat. Sci.* **100** 012017 <https://doi.org/10.1088/1757-899X/100/1/012017>
- [14] Więckowski D, Pusty T and Jędryś P 2016 Influence of the vertical load exerted by the trailer on the coupling device on towing vehicle's steerability and stability *IOP Conf. Ser.-Mat. Sci.* **148** 012031 <https://doi.org/10.1088/1757-899X/148/1/012031>
- [15] Chuan Sun, Jianyu Wang, Lian Xie, Duanfeng Chu and Liqun Liu 2018 Research on Road Safety Evaluation in Curves Based on TruckSim-Simulink Co-simulation *IOP Conf. Ser.-Mat. Sci.* **392** 062157 <https://doi.org/10.1088/1757-899X/392/6/062157>
- [16] Stroganov Yu N, Stroganova O Yu and Ognev O G 2019 Improving design safety of tractor-trailers by upgrading towing couplers *IOP Conf. Ser.-Mat. Sci.* **632** 012028 <https://doi.org/10.1088/1757-899X/632/1/012028>